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Enhancement of the Separate Sensible and Latent Cooling Air-Conditioning Systems

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ABSTRACT

A separate sensible and latent cooling (SSLC) air-conditioning system integrated with a desiccant wheel (where the desiccant is regenerated with the heat from the condenser/gas cooler) provides potential energy savings as compared to a conventional air-conditioning system. The main design challenge is providing sufficient hot air flow rate to the desiccant wheel (DW) for its regeneration, which can easily result in an excessive discharge pressure of the vapor compression system as compared to the conventional system. This paper proposes the idea of applying divided condensers (or gas coolers) to the R410A (or CO_2) SSLC system. To investigate the feasibility of this enhancement option, both the baseline and the SSLC air-conditioning systems were modeled. The models were optimized to achieve maximum COP while maintaining the required air regeneration temperature and cooling capacity. The simulation results show that the application of divided heat exchangers to the SSLC system provides sufficient hot air flow for regenerating the desiccant wheel at both a reduced high side pressure (from 10.4 MPa to 9.7 MPa for CO_2, from 3.46 MPa to 3.45 MPa for R410A) and a reduced discharge temperature from the condenser (gas cooler) (4K lower for both refrigerants) of the vapor compression cycle. For the CO_2 cycle, the simulation results show that the addition of a suction line heat exchanger further improves the performance by 7% on the basis of the SSLC system with divided gas coolers.

Key words: Separate sensible and latent cooling, Divided gas cooler, R410A, CO_2

1. INTRODUCTION

The fact that air-conditioning systems account for over 15% of total household electricity consumption draws extensive studies on how to improve the energy efficiencies of such systems. Separate sensible and latent cooling (SSLC) is viewed as one of the many technologies which can reduce the compressor power input. The reason behind the reduced power input is that the increased refrigerant evaporating pressure of the sensible cooling cycle, which resulted from an increased air temperature, results in a lower pressure ratio across the compressor than that of the conventional one. The reduced pressure ratio brings the benefits of better compressor efficiency and less power input than the conventional system. There are different technologies available to achieve the separation of the sensible and latent cooling. Ling et al. (2009) proposed an idea of using two vapor compression systems for the separation, one of which dealt with sensible load only, while the other one dealt with both latent load and a small amount of sensible load. The energy savings was 30% under the standard condition (35°C, 44% RH), and up to 50% under the hot and dry condition (37°C, 15% RH). More studies were focused on the application of using a vapor compression cycle for sensible load removal and solid/liquid desiccant material for latent load removal. Dai et al. (2001) studied the application of integrating a liquid desiccant and a vapor compression cycle. The test was conducted under the AHRI standard 210/240 (35°C, 40% RH), and the cooling capacity was 5 kW. The COP of the vapor compression cycle improved from 2.2 to 3.39 because of the assistance from liquid desiccant. Dhar and Singh (2001) simulated a hybrid system of a solid desiccant wheel (DW) and a vapor compression cycle. They demonstrated that the hybrid system had a maximum energy savings under hot, dry weather. In the hot and humid region, energy savings was still possible but the latent load should be high. Depending on different desiccant materials, the temperatures for regeneration can be varied from lower than 50°C to above 100°C. Jia et al studied the performance of a solid desiccant wheel which used lithium chloride as the adsorbent. The temperature required to regenerate the wheel was set to be 100°C, and one regeneration heater was used as the heat source. Casas and Schmitz (2004) studied the integration of a desiccant wheel and a cooling, heating, and power (CHP) unit. In their study, the waste heat from the CHP unit could be utilized for the lithium chloride regeneration. However, the regeneration temperature was only in the range of 50°C to 60°C. The difference in regeneration temperatures in the above two papers may be caused by different dehumidification requirements. Silica gel is another widely accepted candidate as desiccant material, and its regeneration temperature is usually higher than 70°C. As new technology emerges, low-temperature regenerated desiccant materials have attracted the attention of many researchers studying the integration of the desiccant wheel and vapor compression cycle because it enables use of the waste heat directly from the condenser or the gas cooler. This paper focuses on using a low-temperature regenerated desiccant wheel for latent load removal, and the desiccant wheel was regenerated by hot air off the condenser (R410A system) or the gas cooler (CO_2 system), namely DW-assisted SSLC system. In order to achieve desired performance, the desiccant wheel was selected to be regenerated at the temperature range of 45°C or higher. In the current study, the regeneration temperature was fixed at 50°C. One of the major challenges for obtaining as much energy savings as possible with this type of system was how to optimize the high side pressure of the vapor compression cycle while balancing the hot discharge air requirement by the desiccant wheel for effective regeneration and suppressing the compression ratio. To address the challenge, we proposed the application of divided condensers (or gas coolers). Instead of heating the entire amount of air through the heat exchanger to the required regeneration temperature, only one section of the heat exchanger (first section) is used for divided heat exchangers as a means to provide hot air for the DW regeneration, while the other section (second section) is used for heat rejection only, without meeting the temperature requirement for the DW regeneration. Therefore, the refrigerant high side pressure is suppressed while the system is still able to effectively regenerate the desiccant wheel. Furthermore, as a common practice to improve the performance of CO_2 cycle, the addition of suction line heat exchanger helps lower the refrigerant discharge temperature off the gas cooler; therefore its integration with the SSLC system was also investigated.

2. SYSTEM MODELING

2.1 System configurations

In this study, the DW-assisted SSLC system has two major components. One component is a vapor compression system, which includes a compressor, an evaporator, an expansion device, and a condenser (or gas cooler). A suction line heat exchanger was exclusively added to the CO_2 vapor compression cycle in this paper. The other component is a desiccant wheel. Figure 1 describes the refrigerant cycle of the DW-assisted SSLC system. The condenser (gas cooler) in the vapor compression cycle is divided into two parts. The refrigerant that is discharged from the compressor enters the first part of the condenser (or gas cooler), and then enters the second part, which is connected to the first one in series. The ambient air flowing to the first part of the heat exchanger is directly sent to the desiccant wheel for regeneration. For the second part of the heat exchanger, the ambient air only serves as the heat sink for the inside refrigerant. The refrigerant leaving from the second part is sent to the expansion device. Figure 2 describes the air flows in the DW-assisted SSLC system. The system has no ventilation air and the return air from the space is 27°C and 50% RH. The return air flows through the evaporator for removing the sensible heat load and splits into two streams in the downstream of the evaporator. One of the streams directly supplies to the space and the other one flows through the desiccant wheel. The desiccant wheel dehumidifies the air stream and the dry air is sent to the space. Each part of the condensers (gas coolers) faces the ambient air of the same condition, which is 35°C and 44% RH.

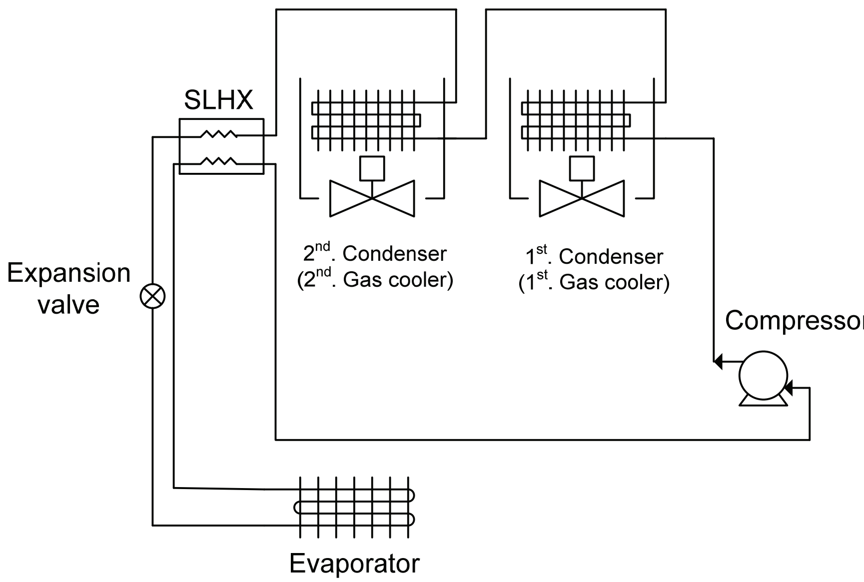

Figure 1: The refrigerant cycle of the DW-assisted SSLC system with divided HXs

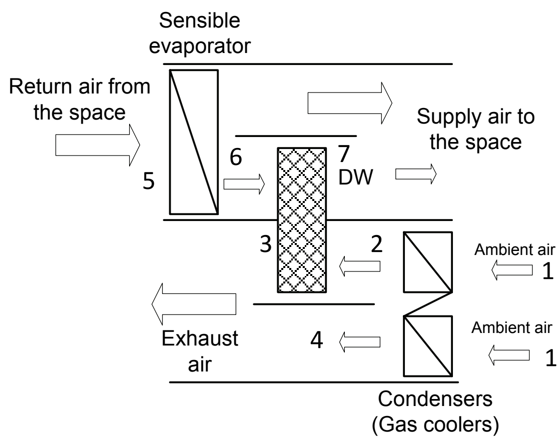

Figure 2: The air cycle of the DW-assisted SSLC system with divided heat exchanger

2.2 System modeling approach

Engineering Equation Solver (EES) was used to model the vapor compression cycle and the desiccant wheel. CoilDesigner, an in-house heat exchanger simulation software package, was used to model all of the heat exchangers in the vapor compression cycle. The built-in optimization tool in EES was utilized to optimize the system COP. The detailed assumptions adopted for calculating the vapor compression cycle were listed as follows:

The integration between EES and CoilDesigner: For each heat exchanger calculation, a database was created by a multiple-variable parametric study in the CoilDesigner. Specifically, the database of evaporator, evaporating pressure, inlet quality and mass flow rate were selected as variables, and for the database of condenser (gas cooler), condensing pressure (high side pressure), inlet temperature, and mass flow rate were selected as variables. Each database had 1,000 (10*10*10) records. EES imported all the records and saved them as 3-dimension arrays. Linear interpolation method was applied to calculate the results from the database records.

Optimization approach: The EES built-in optimization tool was used to maximize the system COP. Because of the nature of the system model, which was non-linear and the possible existence of multiple local maximums, genetic method was applied for the optimization. The optimization function was defined as,

$$Min\ f = -(COP\ of\ the\ system) \tag{1}$$

$$s.t. \qquad system\ capacity \geq 3{,}800\ W\ (SSLC\ system) \tag{2}$$

$$Air\ discharge\ temperature\ off\ the\ 1st\ condenser\ (GC)\ \geq 50℃ \tag{3}$$

The normalized objective function with penalty factor was demonstrated below,

$$f = -\frac{COP}{baseline\ COP} + rp * (\max\left(\frac{-Qeva+3800}{3800}, 0\right) + \max\left(\frac{-t_{airoff}+50}{50}, 0\right)) \tag{4}$$

Where:

Baseline COP = 3, *rp* = 1000

The following assumptions were made for modeling the vapor compression cycle:

- System capacity: 3.5 kW
- Sensible heat factor: 0.7
- Latent capacity: 1.0 kW
- Outdoor/indoor air conditions: 35°C, 44% RH / 27°C, 50% RH
- Refrigerant: R410A and CO_2
- Regeneration temperature: 50°C
- Regeneration air flow rate: 0.15 m^3/s
- Indoor air flow rate: 0.41 m^3/s
- Total air flow rate for condensers (gas coolers): 0.38 m^3/s

Compressor modeling: The compressor's discharge temperature and power input were calculated from compressor isentropic efficiency, compressor efficiency and volumetric efficiency. The three efficiencies were functions of pressure ratio, degree of superheating, compressor frequency and theoretic work input and the functions were obtained through curve fitting from experimental data.

$$\eta_{ise} = f_1(PR, T_{sup}) = \frac{h_{dis,ise} - h_{suc}}{h_{dis} - h_{suc}} \tag{5}$$

$$\eta_{vol} = f_2(PR, T_{sup}, compfreq) = \frac{\dot{m}}{RPM * disp * \rho_{ref}} \tag{6}$$

$$\eta_{comp} = f_3(theoretic\ work\ input) = \frac{theoretic\ work}{real\ work} \tag{7}$$

Heat Exchanger (HX) modeling: all the heat exchanger calculations were conducted in CoilDesigner.

Expansion device: The expansion process in the vapor compression cycle was treated as isenthalpic.

$$h_{in} = h_{out} \tag{8}$$

The desiccant wheel was also modeled in the EES. The dehumidification performance was assumed to be a function of desiccant wheel rotation speed, regeneration temperature, air velocity through the wheel and inlet humidity ratio in the regeneration side. The enthalpy of air in the process side off the desiccant wheel was calculated as a function of desiccant wheel rotation speed, regeneration temperature, and air velocity. All the input data described above were from experimental data.

$$WVR = f_4(T_{reg}, RPH, V_{air}, \omega_{regin}) \tag{9}$$

$$h_{gain} = f_5(T_{reg}, RPH, V_{air}) \tag{10}$$

Suction line heat exchanger: The SLHX was modeled as a tube-in-tube heat exchanger in the CoilDesigner. The tube inner and outer diameters were chosen in order to satisfy two requirements. The first requirement was the ability to withhold the inside refrigerant of high side pressure and the second requirement was the ability to maintain the pressure drop of less than 1% of the evaporating pressure, which was recommended by S.A. Klein et al. The tube length was adjusted to make the efficiency of the SLHX reach 0.8. Table 1 shows the different designs of the SLHX by the different efficiencies.

Table 1: The designs of the SLHX

SLHX efficieny (-)	Capacity (kW)	High side pressure drop (kPa)	Low side pressure drop (kPa)	Tube length (m)	Outer tube inner diameter (mm)	Inner tube diameter outer/inner (mm)
0.0	0.00	0.0	0.0	0.0	12	6.3/4.3
0.2	0.19	4.6	2.5	0.4	12	6.3/4.3
0.4	0.37	11.5	6.4	1.0	12	6.3/4.3
0.6	0.55	22.8	13.1	1.9	12	6.3/4.3
0.8	0.74	47.8	28.4	4.0	12	6.3/4.3

3. MODELING RESULTS AND DISCUSSION

Tables 2 and 3 provide validation of the EES-CoilDesigner models by comparing the simulation results with the experimental data. The range of pressures in those tables is small because both the refrigerant evaporating temperature and the air temperature discharging from the first section of the condenser (or gas cooler) maintained at 20°C and 50°C in the experiments. The largest discrepancy (5%) between the simulations and experiments comes from modeling of the R410A compressor. It indicates that the three efficiencies based modeling method has a limited accuracy. However, because it is only 5%, no further modifications to the models were attempted. The detailed design of divided heat changers is shown in Table 4. The total volume and heat transfer area of the two divided heat exchangers maintain the same as those of a single heat exchanger. The single heat exchanger in the SSLC system has a smaller air flow rate than the divided heat exchangers and the baseline one, which is because of the regeneration temperature requirement of the discharge air. The amount of air flow through the first section of the divided gas cooler was determined by the required latent capacity (1 kW), and the total air flow rate of the two divided heat exchangers was set to be the same as that of the baseline system. The division of the two heat exchangers satisfies the requirement that the air velocities are the same in both sections and baseline system. Figure 3 demonstrates the psychrometric process of the DW-assisted SSLC system with divided condensers (gas coolers). Different line styles represent different air processes. The bold numbers in Figure 3 represent air state points corresponding to ones in their respective schematic drawing (Figure 2). For the comparison purpose, two baseline systems were modeled. One of them was a conventional air conditioning system, which had four major components of the vapor compression cycle (compressor, evaporator, expansion device and condenser/gas cooler). The vapor compression system removes both the sensible and latent loads of the space. The capacity and the sensible heat factor (SHF) were set to be the same as those of the SSLC systems (3.5 kW, SHF=0.7). The other one is the DW-assisted SSLC system using a single condenser (or gas cooler) without dividing it.

Table 2: Validation of the R410A SSLC system modeling

	Input			Output		
	Cond. inlet pressure (kPa)	Evap. inlet pressure (kPa)	Refrigerant MFR (g/s)	Evap. capacity (kW)	Cond. capacity (kW)	Power input (kW)
EXP	3422.7	1354.1	23.6	3.81	4.36	0.85
EES-CD				3.81	4.41	0.84
EXP	3417.8	1341.3	23.1	3.81	4.33	0.85
EES-CD				3.75	4.34	0.83
EXP	3445.0	1337.0	22.3	3.70	4.23	0.83
EES-CD				3.63	4.21	0.81
EXP	3442.8	1345.1	21.7	3.62	4.13	0.81
EES-CD				3.55	4.11	0.77

Table 3: Validation of the CO_2 SSLC system modeling

	Input			Output		
	GC inlet pressure (kPa)	Evap. inlet pressure (kPa)	Refrigerant MFR (g/s)	Evap. capacity (kW)	GC capacity (kW)	Power input (kW)
EXP	10969.2	5395.9	30.4	3.91	4.65	1.19
EES-CD				3.79	4.55	1.18
EXP	10813.7	5437.2	32.1	3.98	4.73	1.20
EES-CD				3.87	4.63	1.20
EXP	10438.8	5615.5	38.4	4.03	4.73	1.23
EES-CD				3.98	4.81	1.25
EXP	10931.3	5479.8	31.5	3.88	4.61	1.19
EES-CD				3.84	4.59	1.17

Table 4: The designs of the divided heat exchangers and the single heat exchanger

Specifications \ HXs	Single GC/condenser	Divided high temp. HX	Divided low temp. HX
Number of banks	4	4	4
Number of circuits	2	2	2
Number of tubes per bank	22	8	14
Air VFR (m^3/s)	0.249	0.149	0.224
Frontal area (mm×mm)	H 484 × W 622	H 176 × W 622	H 308 × W 622
Tube OD (mm)	7.94	7.94	7.94
Wall thickness (CO_2) (mm)	1.00	1.00	1.00
Wall thickness (R410A) (mm)	0.31	0.31	0.31

In Figure 4, the COPs of the SSLC systems aforementioned are compared with the baseline COPs. The system COP was defined as the ratio of the space cooling capacity (3.5 kW) to the compressor power input. For the baseline systems, the system COP could be considered same as the vapor compression cycle COP, which is defined as the ratio of the evaporator air side capacity to the compressor power input. However, for the DW-assisted SSLC system, the vapor compression cycle usually had to provide a greater amount of cooling in order to maintain the space cooling capacity. It is because the real dehumidification process of the desiccant is not isenthalpic, and it generates extra heat to the system. As shown in Figure 4, the conventional systems had COP of 3.7 and 2.7 for the R410A and CO_2 systems, respectively. The DW-assisted SSLC system successfully separated the latent load from the vapor compression cycle, but the COP improvements were only 21.2% and 20.1%, for the R410A and CO_2 systems, respectively. There are two reasons affecting the improvements. First of all, in order to provide 50°C air for regeneration, the amount of air flowing through the condenser or gas cooler was less than those in the conventional system (0.25 m^3/s vs. 0.37 m^3/s), which led to a higher refrigerant discharge temperature off the condenser (gas cooler) and therefore, a higher approach temperature. The pressure of refrigerant was also kept at a high level, which was 10.4 MPa for the CO_2 system, and 3.46 MPa for the R410A system. Both of them were higher than those in the baseline systems. The second reason came from the extra heat provided to the system caused by the desiccant wheel. The vapor compression cycles in the DW-assisted SSLC systems had to deliver almost 10% higher capacity than those in the conventional systems (3.8 kW vs. 3.5 kW), which made the HXs in the SSLC system undersized compared with those in the conventional systems. The second problem was difficult to solve since it primarily came from the heat of adsorption generated by the adsorbent. However, the first problem could be solved by the application of dividing HXs. The CO_2 vapor compression cycles were plotted in the P-h diagram as shown in Figure 5. The cycle in dotted line is for the DW-assisted SSLC system without gas cooler being divided, and the cycle in solid line is the one with divided gas coolers. It clearly shows that the application of divided heat exchangers reduce the high side pressure and also reduce the approach temperature. In more detail, the divided gas coolers reduce the high side pressure from 10.4 MPa to 9.7 MPa, and the refrigerant temperature off the gas cooler is reduced from 42.2°C to 38.0°C. For the R410A system, the pressure reduction is insignificant (3.46 MPa to 3.45 MPa); however, the discharge temperature reduction is significant (from 40°C to 36 °C). The hollow legends in Figure 4 represent the cases of CO_2 SSLC systems with the addition of the suction line heat exchanger. In both cases, the improvements of COP are 7%. These improvements come from further reduced high side pressure and refrigerant discharge temperature off the condenser (gas cooler). In the SSLC with single heat exchanger, the SLHX makes the high side pressure reduce from 10.4 MPa to 10.3 MPa, and the discharge temperature reduce from 42.2°C to 41.8°C, while in the SSLC with divided heat exchanger, the reductions are from 9.7 MPa to 9.5 MPa and from 38.0°C to37.7°C. However, the tradeoff from the addition of SLHX is the increased compressor discharge temperature. In both cases, the discharge temperatures increase by 12 K and 4 K. Compared with the R410A baseline system's COP of 3.67, the R410A SSLC system with divided condensers had a COP of 5, which provided 36% improvement. CO_2 cycle responds better to the divided heat exchangers in terms of COP improvement than the R410A cycle. Its baseline system had a COP of 2.7 and was improved to 4.35 by the application of divided gas coolers. It can be found that the reduction of high side pressure of CO_2 cycle is more significant than the one of R410A cycle. It is probably because the divided gas coolers have a better effect of reducing the approach temperature between the air and the CO_2 By reducing the fin conduction effect.

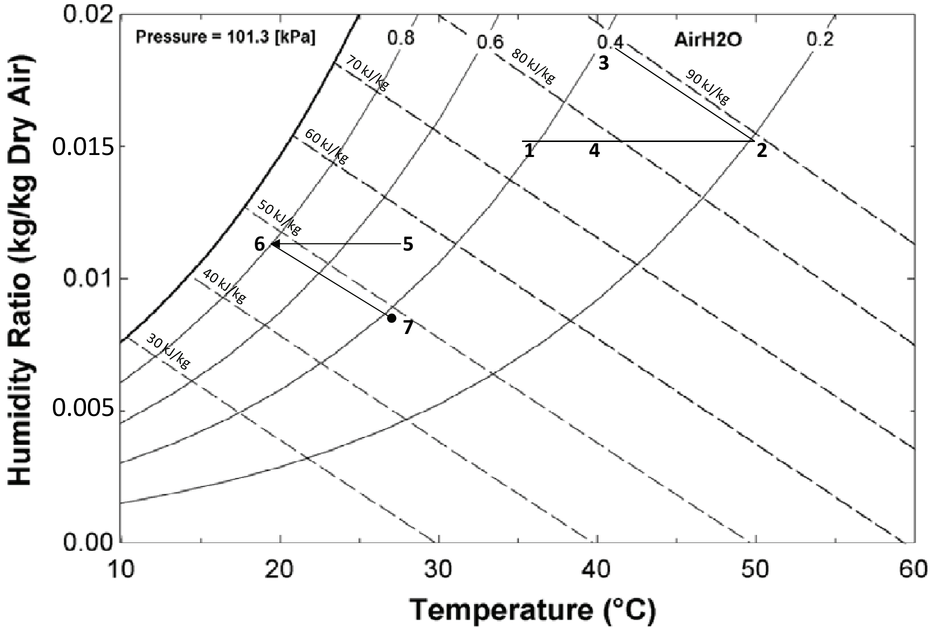

Figure 3: Psychrometric process of DW-assisted SSLC system with divided HXs

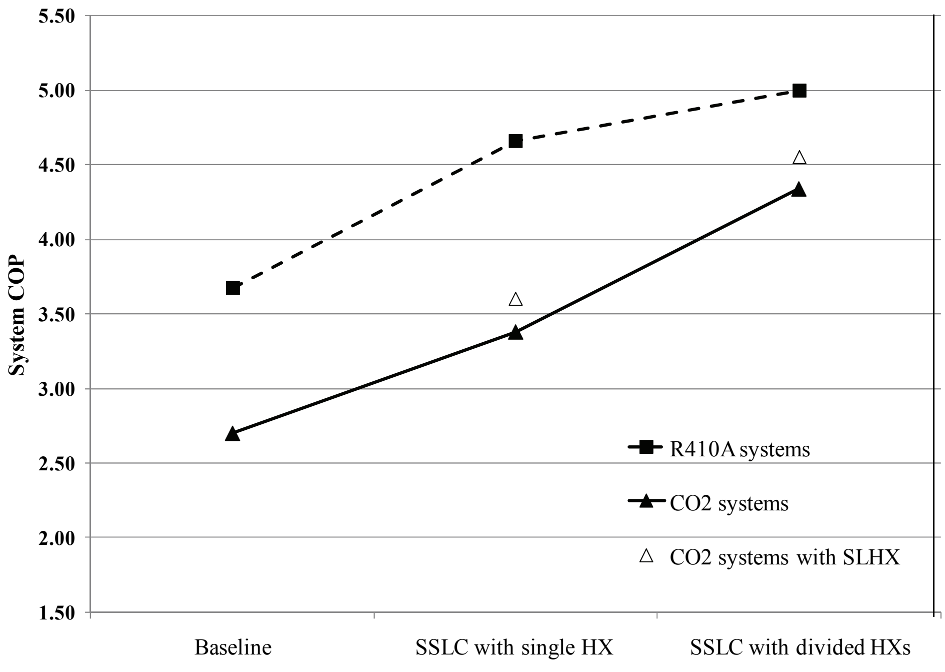

Figure 4: The COP of different SSLC options and baseline system

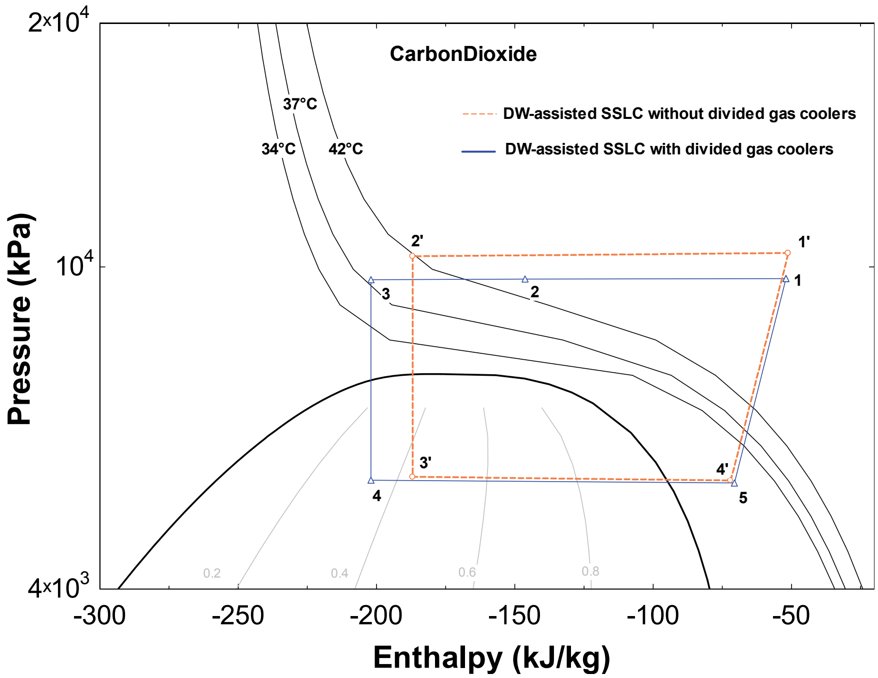

Figure 5: The P-h diagram of CO_2 DW-assisted SSLC systems

(1': compressor outlet, 2': gas cooler outlet, 3': expansion device outlet, 4': evaporator outlet

1: compressor outlet, 2: 1st gas cooler outlet, 3: 2nd gas cooler outlet, 4: expansion device outlet, 5: evaporator outlet)

4. CONCLUSIONS

This paper focuses on the enhancement of the DW-assisted SSLC system. The integration of EES and CoilDesigner, which is specialized in heat exchangers simulation, helps improve the accuracy of the models. The models were validated by comparing the simulation results with the experimental data. The methods to further enhance the DW-assisted SSLC system were hence theoretically studied. The idea of dividing condensers (or gas coolers) resulted in a reduced high side pressure as well as the refrigerant discharge temperatures. The SSLC with divided condensers (or gas coolers) led to the COP improvement of 36% and 61% for R410A and CO_2 systems compared to the baseline counterparts, respectively. The addition of SLHX to the CO_2 SSLC system provided another 7% of improvement on the basis of dividing gas coolers. The CO_2 system responded better to the divided heat exchangers than the R410A system by minimizing fin conduction effect.

NOMENCLATURE

cond	condenser	
COP	coefficient of performance	
disp	compressor displacement volume	(cc/rev.)
DOS	dedicated outdoor system	
DW	desiccant wheel	
EES	engineering equation solver	

evap	evaporator	
exp	experiment	
GC	gas cooler	
h	enthalpy	(kJ/kg)
$\dot{m}$	mass flow rate	(g/s)
SSLC	separate sensible and latent cooling	
Ref	refrigerant	
RH	relative humidity	
RPH	desiccant wheel rotation speed	(rev. per hour)
RPM	compressor rotation speed	(rev. per min.)
rp	penalty factor	
PR	pressure ratio	
Q	capacity	(kW)
WVR	water vapor removal rate	(g/s)
η	efficiency	

Subscripts

comp	compressor
dis	discharge
in	inlet
ise	isentropic
out	outlet
vol	volumetric

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